CHAPTER 3

SEALS AND GASKETS

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3.1 INTRODUCTION

A seal is a device placed between two surfaces to prevent the flow of gas or liquid from one region to another. Seals are used for both static and dynamic applications. Static seals such as gaskets, bolt seals, back-up rings and sealants are used to prevent leakage through a mechanical joint when there is no relative motion of mating surfaces. Truly static seals are designed to provide a complete barrier to a potential leakage path. These seals are "zero leakage" seals (down to 10^{-11} scc/sec.helium). In a truly static seal, the mating gland parts are not subject to relative movement except for thermal expansion and movement from the application of fluid pressure. Some static seals are designed to accommodate limited movement of the surfaces being sealed due to changes in pressure, vibration or thermal cycling such as an expansion joint. These seals are sometimes referred to as semi-static seals.

A dynamic seal is a mechanical device used to control leakage of fluid from one region to another when there is rotating, oscillating or reciprocating motion between the sealing interfaces. An O-ring can be used in both static and dynamic applications. However, the employment of O-rings as primary dynamic seals is normally limited to short strokes and moderate pressures. An example of static and dynamic seal applications is shown in Figure 3.1.

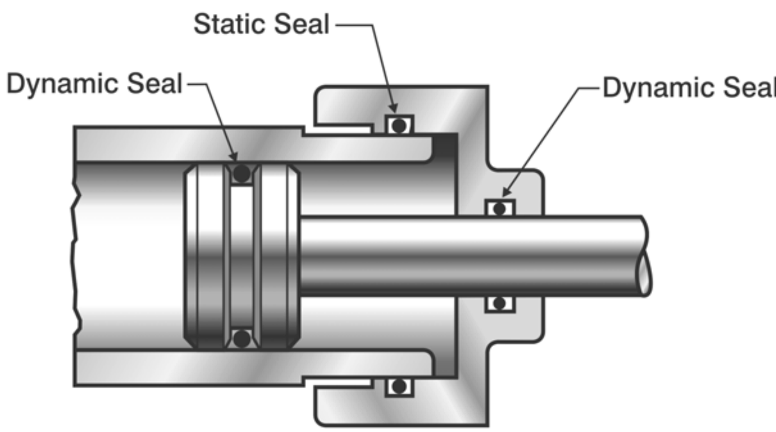


Figure 3.1 Static and Dynamic Seals

Other seal designs include reciprocating seals, oscillating seals and rotary seals where movement relative to other mechanical parts will occur. Reciprocating seals involve relative reciprocating motion along the shaft axis between the inner and outer elements. In reciprocating seal applications, the O-ring slides or rocks back and forth within its gland with the reciprocating motion. Reciprocating seals are most often seen in cylinders and linear actuators. In oscillating seal applications, the inner or outer member of the gland moves in an arc around the axis of the shaft - first in one direction and then in the opposite direction, generally intermittently with no more than a few turns in each direction. The most common application for oscillating O-ring seals is in faucet valves. Rotary seals involve motion between a shaft and a housing. Typical rotary seals include motor shafts and wheels on a fixed axle. These specific seal designs are included in the appropriate section of this chapter as either static of dynamic seals.

A mechanical seal is designed to prevent leakage between a rotating shaft and its housing under conditions of higher fluid pressure, shaft speed and temperature normally associated with dynamic seals. For purposes of this Chapter, a mechanical seal will be assumed to be rotating in contact with the fluid above 800 rpm or a sliding contact exceeding 600 feet/minute. A contact sealing face composed of a soft, sacrificial face material forms a seal against a hard material. A common design has a carbon rotating element. Failure of a mechanical seal is defined as an inoperative seal before wear-out of the sacrificial surface.

The reliability of a seal design is determined by the ability of the seal to restrict the flow of fluid from one region to another for its intended life in a prescribed operating environment. The evaluation of a seal design for reliability must include a definition of the design characteristics and the operating environment in order to estimate its design life. Section 3.2 discusses the reliability of gaskets and other static seals. Procedures for evaluating the reliability of dynamic seals are contained in Section 3.3. Procedures for evaluating mechanical seals for reliability are included in Section 3.4.

3.2 GASKETS AND STATIC SEALS

A gasket is used to develop and maintain a barrier between mating surfaces of mechanical assemblies when the surfaces do not move relative to each other. The barrier is designed to retain internal pressures, prevent liquids and gases from escaping the assembly, and prevent contaminants from entering the assembly. Gaskets can be metallic or nonmetallic. Flange pressure compresses the gasket material and causes the material to conform to surface irregularities in the flange and is developed by tightening bolts that hold the assembly together.

Gasket reliability is affected by the type of liquid or gas to be sealed, internal pressure, temperature, external contaminants, types of surfaces to be joined, surface roughness, and flange pressure developed at the joint. To achieve the barrier to a potential leakage path the seal must be sufficiently resilient to conform to cavity irregularities and imperfections, while remaining rigid enough to provide the required contact force needed to ensure a tight seal. This contact force is a function of the seal cross section, as well as the compression of the seal between the mating cavity faces. The load on the gasket must be distributed evenly over the whole area of the gasket rather than have a few points of high load with reduced stress at midpoints between the fasteners. Therefore, a larger number of small bolts is better than a few larger bolts. Use of a torque wrench during installation is always a necessity.

An O-ring is a mechanical gasket in the shape of a torus. It is a loop of elastomer with a disc-shaped cross-section, designed to be seated in a groove and compressed during assembly between two or more parts, creating a seal at the interface. The combination of the O-ring and the gland that supports the O-ring constitute the classic O-ring seal assembly. A typical O-ring configuration is shown in Figure 2.

While static seals in most cases are designed for “zero-leakage” semi-static seals for applications where there is limited movement are not designed or intended to be “zero-leakage” seals. Their contact or compression force is typically an order of magnitude lower than a static seal.

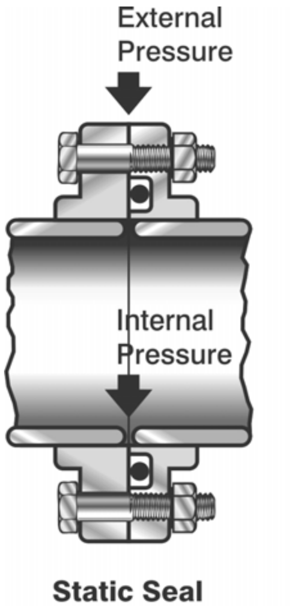


Figure 3.2 Typical O-ring Configuration

3.2.1 Static Seal Failure Modes

The primary failure mode of a gasket or static seal is leakage. In the case of an O-ring, the O-ring flows up to, but not into, the clearance gap between components under normal pressure. If the pressure is increased, both the sealing force and contact area increase. At the seal pressure limit depending on the seal material and hardness, part of the O-ring starts to extrude into the clearance gap. At this point the seal can shear creating seal leakage.

Static seals may also be subjected to high and low temperatures, chemical attack, vibration, abrasion, and movement. The integrity of a seal depends upon the compatibility of the fluid and sealing components, conditions of the sealing environment, and the applied load during application.

A failure mode especially applicable to low pressure applications is compression set. Compression set refers to the permanent deflection remaining in the seal after complete release of a squeezing load while exposed to a particular temperature level. Compression set reflects the partial loss of elastic memory due to the time effect. Operating over extreme temperatures can result in compression-type seals such as gaskets and O-rings to leak fluid at low pressures because they have deformed permanently or taken a set after used for a period of time.

All seals have an upper temperature limit determined by the type and grade of the material being used. Thermal expansion can cause misalignment and cause uneven surfaces that are designed to remain flat. Static seals also undergo plastic deformation during installation and maintenance procedures need to be reviewed for those situations where permanent set can take place before permitting the seal to be replaced. Table 3-1 contains a list of typical failure mechanisms and causes of seal leakage. Other failure mechanisms and causes should be identified for the specific product to assure that all considerations of reliability are included in any design evaluation.

3.2.2 Failure Rate Model Considerations

A review of failure rate data suggests the following characteristics be included in the failure rate model for gaskets and seals:

- Material characteristics
- Amount of seal compression
- Surface irregularities
- Seal size

- Fluid pressure
- Extent of pressure pulses
- Temperature
- Fluid viscosity
- Contamination level
- Fluid/material compatibility
- Leakage requirements
- Assembly/quality control procedures

Table 3-1. Typical Failure Mechanisms and Causes for Static Seals and Gaskets

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Leakage	- Wear	- Contaminants - Misalignment - Vibration - Poor surface finish
	- Elastic Deformation - Gasket/seal distortion	- Extreme temperature - Misalignment - Seal eccentricity - Extreme loading / extrusion - Compression set/overtorqued bolts
	- Surface Damage - Embrittlement	- Inadequate lubrication - Contaminants - Fluid/seal degradation - Thermal degradation - Idle periods between component use - Exposure to atmosphere, ozone - Excessive temperature
	- Creep	- Fluid pressure surges - Material degradation - Thermal expansion & contraction
	- Compression Set	- Excessive squeeze to achieve seal - Incomplete vulcanization - Hardening/high temperature
	- Installation Damage	- Insufficient lead-in chamfer - Sharp corners on mating metal parts - Inadequate protection of spares
	- Gas expansion rupture	- Absorption of gas or liquefied gas under high pressure

The failure rate of a static seal is a function of actual leakage and the allowable leakage under conditions of usage, failure occurring when the rate of leakage reaches a predetermined threshold. This rate, derived empirically, can be expressed as follows:

$$\lambda_{SE} = \lambda_{SE,B}\left(\frac{Q_a}{Q_f}\right) \tag{3-1}$$

Where: λ_{SE} = Failure rate of gasket or seal considering operating environment, failures per million hours

$\lambda_{SE,B}$ = Base failure rate of seal or gasket due to random cuts, installation errors, etc. based on field experience data, failures per million hours

Q_a = Actual leakage rate, in^3/min

Q_f = Allowable leakage rate under conditions of usage, in^3/min

Allowable leakage is dependent on the application. External leakage of a component containing water is obviously not as critical as one containing fuel. Allowable leakage, Q_f is determined from design drawings, specifications or knowledge of component applications. The actual leakage rate Q_a for a seal is determined from the standard equation for laminar flow around two curved surfaces (Reference 5):

$$Q_a = \left(\frac{\pi\left(P_1^2 - P_2^2\right)}{25\,\nu_a P_2}\right)\left(\frac{r_o + r_i}{r_o - r_i}\right) H^3 \tag{3-2}$$

Where

P_1 = System or upstream pressure, lbs/in^2

P_2 = Standard atmospheric pressure or downstream pressure, lbs/in^2

ν_a = Absolute fluid viscosity, $lb\text{-}min/in^2$

r_i = Inside radius of circular interface, in

r_o = Outside radius of circular interface, in

H = Conductance parameter, in [See Equation (3-4)]

For flat seals or gaskets the leakage can be determined from the following equation:

$$Q_a = \left(\frac{\pi \; L \left(P_1^{\,2} - P_2^{\,2} \right)}{12 \, \nu_a \; w \, P_2} \right) H^3 \tag{3-3}$$

Where: w = Width of non-circular flat seals, in

L = Contact length, in

The conductance parameter H is dependent upon contact stress of the two sealing surfaces, hardness of the softer material and surface finish of the harder material (Reference 5). First, the contact stress (load/area) is calculated and the ratio of contact stress to Meyer hardness of the softer interface material computed. The surface finish of the harder material is then determined. The conductance parameter is computed from the following empirically derived formula:

$$H = 0.23 \left(\frac{M}{C} \right)^{1.5} \bullet \, f^{\,2/3} \tag{3-4}$$

Where: M = Meyer hardness (or Young's modulus) for rubber and resilient materials, lbs/in^2

C = Contact stress, lbs/in^2 [See Equation (3-9)]

f = Surface finish, in

Seal wear is dependent on the finish of the surface against which the seal rubs when pressure is applied and released or pressure surges occur. The surface finish, f, will deteriorate as a function of time at a rate dependent upon several factors:

- Seal degradation
- Contaminant wear coefficient (in^3/particle)
- Number of contaminant particles per in^3
- Flow rate, in^3/min
- Ratio of time the seal is subjected to contaminants under pressure
- Temperature of operation, ^oF

Note that surface finish and seal hardness are the two parameters in the seal reliability equation that will change as a function of time. Therefore, estimating the value of these parameters at different time intervals during the life of the product will provide an estimate of the total assembly as a function of time.

The contaminant wear coefficient is an inherent sensitivity factor for the seal or gasket based upon performance requirements. The quantity of contaminants includes those produced by wear and ingestion in components upstream of the seal and after the filter. Combining and simplifying terms provides the following equations for the failure rate of a seal.

For circular seals:

$$\lambda_{SE} = \lambda_{SE,B}\left[\frac{K_1\left(P_1^2 - P_2^{\,2}\right)H^3}{Q_f\,\nu_a\,P_2}\right] \bullet \left[\frac{r_o + r_i}{r_o - r_i}\right] \tag{3-5}$$

and, for flat seals and gaskets:

$$\lambda_{SE} = \lambda_{SE,B}\left[\frac{K_2\left(P_1^2 - P_2^{\,2}\right)L\,H^3}{Q_f\,\nu_a\,w\,P_2}\right] \tag{3-6}$$

Where K_1 and K_2 are empirically derived constants

3.2.3 Failure Rate Model for Gaskets and Static Seals

By normalizing the equation to those values for which historical failure rate data from the Navy Maintenance and Material Management (3-M) system are available, the following model can be derived:

$$\lambda_{SE} = \lambda_{SE,B} \bullet C_P \bullet C_Q \bullet C_{DL} \bullet C_H \bullet C_F \bullet C_\nu \bullet C_T \bullet C_N \tag{3-7}$$

Where:

λ_{SE} = Failure rate of a seal in failures/million hours

$\lambda_{SE,B}$ = Base failure rate of seal, 2.4 failures/million hours

C_P = Multiplying factor which considers the effect of fluid pressure on the base failure rate (Figure 3.10)

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.11)

C_{DL} = Multiplying factor which considers the effect of seal size on the base failure rate (See Figure 3.12 for seals or Figure 3.13 for gaskets)

C_H = Multiplying factor which considers the effect of contact stress and seal hardness on the base failure rate (See Figure 3.14)

C_F = Multiplying factor which considers the effect of seat smoothness on the base failure rate (See Figure 3.15)

C_ν = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3)

C_T = Multiplying factor which considers the effect of temperature on the base failure rate (See Figure 3.16)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4)

The parameters in the failure rate equation can be located on an engineering drawing, by knowledge of design standards or by actual measurement. Design parameters other than the above multiplying factors which have a minor effect on reliability are included in the base failure rate as determined from field performance data. The following paragraphs provide background information on those parameters included in the model.

3.2.3.1 Fluid Pressure

Figure 3.10 provides fluid pressure multiplying factors for use in the model. Fluid pressure on a seal will usually be the same as the system pressure. The fluid pressure at the sealing interface required to achieve good mating depends on the resiliency of the sealing materials and their surface finish. It is the resilience of the seal which insures that adequate sealing stress is maintained while the two surfaces move in relation to one another with thermal changes, vibration, shock and other changes in the operating environment. The reliability analysis should include verification that sufficient pressure will be applied to affect a good seal.

At least three checks should be made to assure the prevention of seal leakage:

(1) One surface should remain relatively soft and compliant so that it will readily conform to the irregularities of the harder surface

(2) Sufficient sealing load should be provided to elastically deform the softer of the two sealing surfaces

(3) Sufficient smoothness of both surfaces is maintained so that proper mating can be achieved

3.2.3.2 Allowable Leakage

Figure 3.11 provides an allowable leakage multiplying factor for use in Equation 3-7. Determination of the acceptable amount of leakage which can be tolerated at a seal interface can usually be obtained from component specifications. The allowable rate is a function of operational requirements and the rate may be different for an internal or external leakage path.

3.2.3.3 Seal Size

Figure 3.5 shows a typical installation for a seal and the measurements for r_i and r_o. For a gasket, the inside perimeter dimension *w* and the contact length *L* are used in the equation. Figures 3.12 and 3.13 show the effect of seal size on reliability. The inside diameter of the seal is used in Figure 3.12 as a close approximation of the seal size.

3.2.3.4 Conductance Parameter

Three factors comprise the conductance parameter:

(1) Hardness of the softer material

(2) Contact stress of the seal interface

(3) Surface finish of the harder material

(1) Hardness of the softer material: - In the case of rubber gaskets and O-rings, the hardness of rubber is measured either by durometer or international hardness methods. Both hardness test methods are based on the measurement of the penetration of a rigid ball into a rubber specimen. Throughout the seal/gasket industry, the Shore A durometer is the standard instrument used to measure the hardness of rubber compounds. The durometer has a calibrated spring which forces an indentor point into the test specimen against the resistance of the rubber. The scale of hardness is from 0 degrees for elastic modulus of a liquid to 100 degrees for an infinite elastic modulus of a material, such as glass. Readings in International Rubber Hardness Degree (IRHD) are comparable to those given by a Shore A durometer (Reference18) when testing standard specimens per the ASTM methods. The relationship between the rigid ball penetration and durometer reading is shown in Figure 3.3.

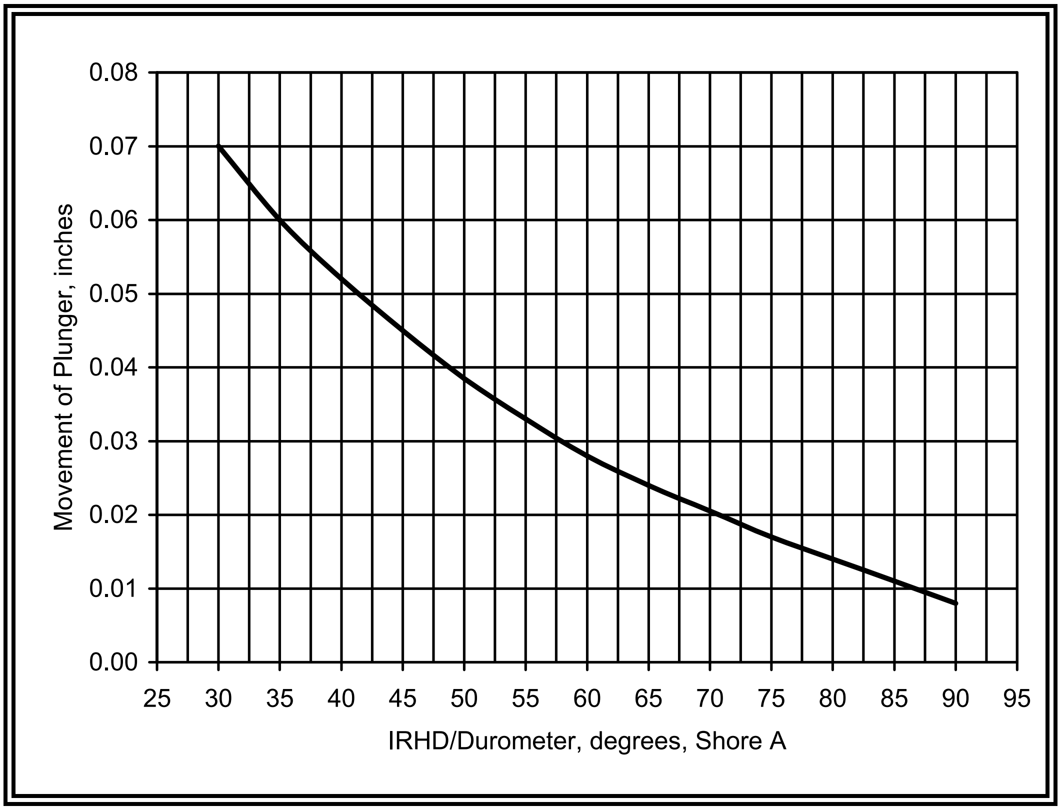


Figure 3.3 Relation Between International Rubber Hardness Degree (IRHD) and Rigid Ball Penetration

Well-vulcanized elastic isotropic materials, like rubber seals manufactured from natural rubbers and measured by IRHD methods, have a known relationship to Young's modulus. The relation between a rigid ball penetration and Young's modulus for a perfectly elastic isotropic material is (Reference 18):

$$\frac{F_1}{M_P} = 1.90\ R_p{}^2 \left(\frac{P_D}{R_P}\right)^{1.35} \tag{3-8}$$

Where:

F_1 = Indenting force, lbf

M_p = Young's modulus, lbs/in^2

R_P = Radius of ball, in

P_D = Penetration, in

Standard IRHD testers have a ball radius of 0.047 inches with a total force on the ball of 1.243 lbf. Using these testing parameters, the relationship between seal hardness and Young's modulus is shown in Figure 3.4. Since Young's modulus is expressed in lbs/in^2 and calculated in the same manner as Meyer's hardness for rigid material; then, for rubber materials, Young's modulus and Meyer's hardness can be considered equivalent.

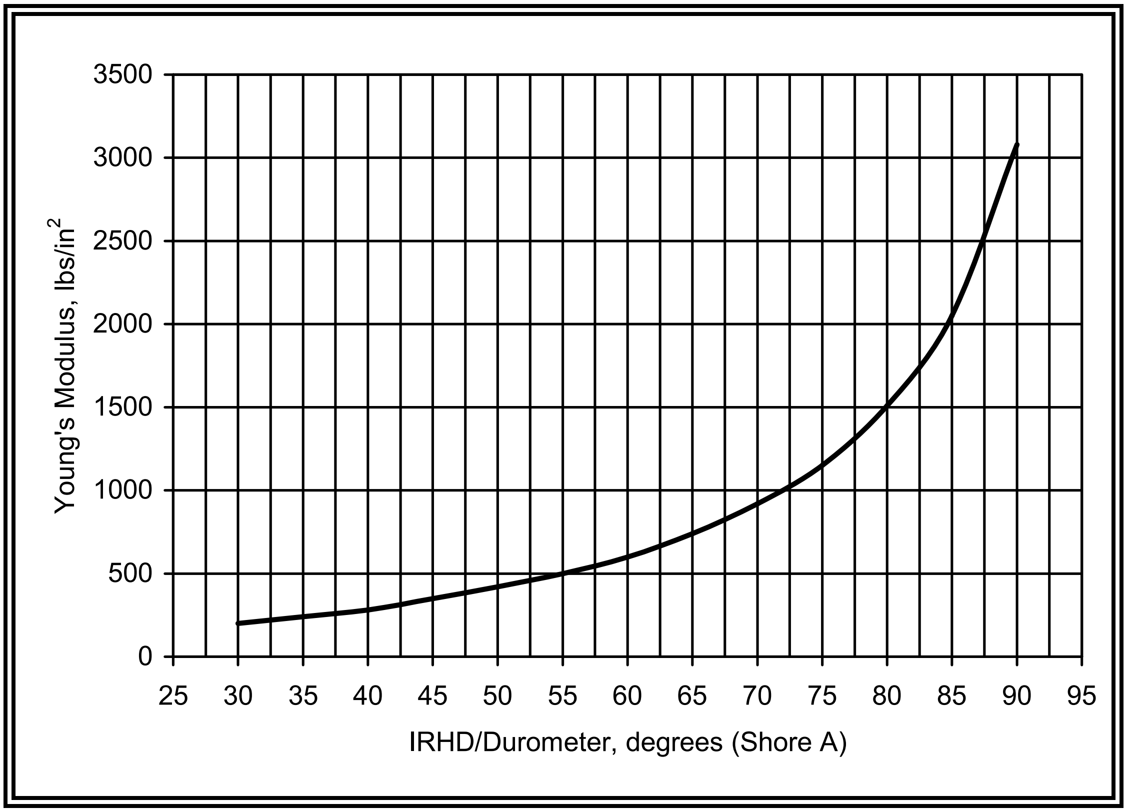


Figure 3.4 Seal Hardness and Young's Modulus

(2) <u>Surface finish of the harder material:</u> - For a gasket, the surface finish of the two surfaces containing the gasket govern the thickness and compressibility necessary for the gasket material for completing a physical barrier in the clearance gap between the flanges. Flatness of the surfaces being sealed is an important consideration. Reliability of the gasket is dependent on the type of material for the specific fluid and application. For seals contained in a gland, as pressure is applied to the fluid component, the O-ring will tend to roll in the gland and a gasket will tend to move between the retaining hardware. The surface finish on the gland will usually be about 32 microinches for elastomer seals, 16 microinches for plastic seals and 8 microinches for metals. In addition to average surface finish, the allowable number and magnitude of flaws in the gland must be considered in projecting leakage characteristics. Flaws such as surface cracks, ridges or scratches will have a detrimental effect on seal leakage. When projecting seal and gasket failure rates for different time periods of the

equipment life cycle, it is important to consider the exposure to contaminants and their effect on surface finish.

(3) Contact stress of the seal interface: - Seals deform to mate with rigid surfaces by elastic deformation. Since the deformation of the seal is almost entirely elastic, the initially applied seating load must be maintained. Thus, a load margin must be applied to allow for strain relaxation during the life of the seal yet not to the extent that permanent deformation takes place. An evaluation of cold flow characteristics is required for determining potential seal leakage of soft plastic materials. Although dependent on surface finish, mating of metal-to-metal surfaces generally requires a seating stress of two to three times the yield strength of the softer material. Figure 3.5 shows a typical installation of a gasket seal.

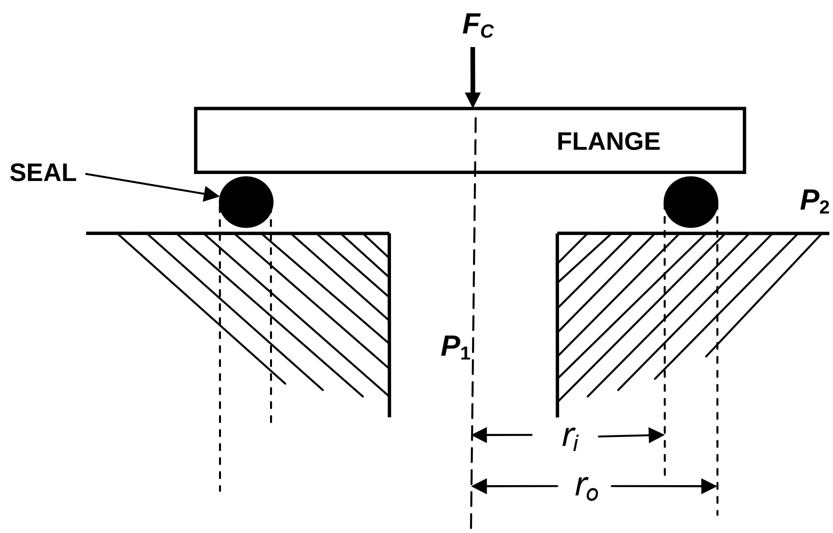


Figure 3.5 Typical Seal Installation

If the seal is pressure energized, the force F_C applied to the seal must be sufficient to balance the fluid pressure forces acting on the seal and thus, prevent separation of the interface surfaces. This requirement is determined by the maximum applied fluid pressure, geometry of the seal groove and pressure gradient at the interface due to leakage. Motion at the interface is prevented by the radial friction forces at the interface to counter the fluid pressure forces tending to radially deform the seal. Thus, the radial restraining force F_C will be greater than the radial pressure deformation forces.

The contact stress, C, in lbs/in^2 can be calculated by:

$$C = \frac{F_C}{A_{SC}} \tag{3-9}$$

Where: F_C = Force compressing seals, lb

A_{SC} = Area of seal contact, in^2

or:

$$C = \frac{F_C - P_1 \pi r_i^2 - (P_1 - P_2)\left(\frac{r_o + r_i}{2}\right)(r_o - r_i)}{\pi\left(r_o^2 - r_i^2\right)} \qquad \textbf{(3-10)}$$

Where: P_1 = System pressure, lbs/in^2

P_2 = Standard atmospheric pressure or downstream pressure, lbs/in^2

r_o = Outside seal radius, in

r_i = Inside seal radius, in

For most seals, the force compressing the seal F_C is normally two and one-half times the Young's modulus for the material. If too soft a material is used, the seal material will have insufficient strength to withstand the forces induced by the fluid and will rapidly fail by seal blowout. If the seal is too hard it will not sufficiently deform in the gland and immediate leakage will occur.

3.2.3.5 Fluid Viscosity

Viscosity of a fluid is much more dependent on temperature than it is on pressure. For example, when air pressure is increased from 1 atmosphere to 50, its viscosity is only increased by about 10%. In contrast, Figure 3.6 shows the dependence of viscosity on temperature for some common fluids. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for the effect of fluid viscosity on the base failure rate of seals and gaskets are provided in Table 3-3. Viscosities for other fluids at the operating temperature can be found in referenced sources and the corresponding multiplying factor determined using the equation following Table 3-3. If the value located is in terms of kinematic viscosity, multiply the value by the specific gravity (density) at the desired temperature to determine the dynamic viscosity.

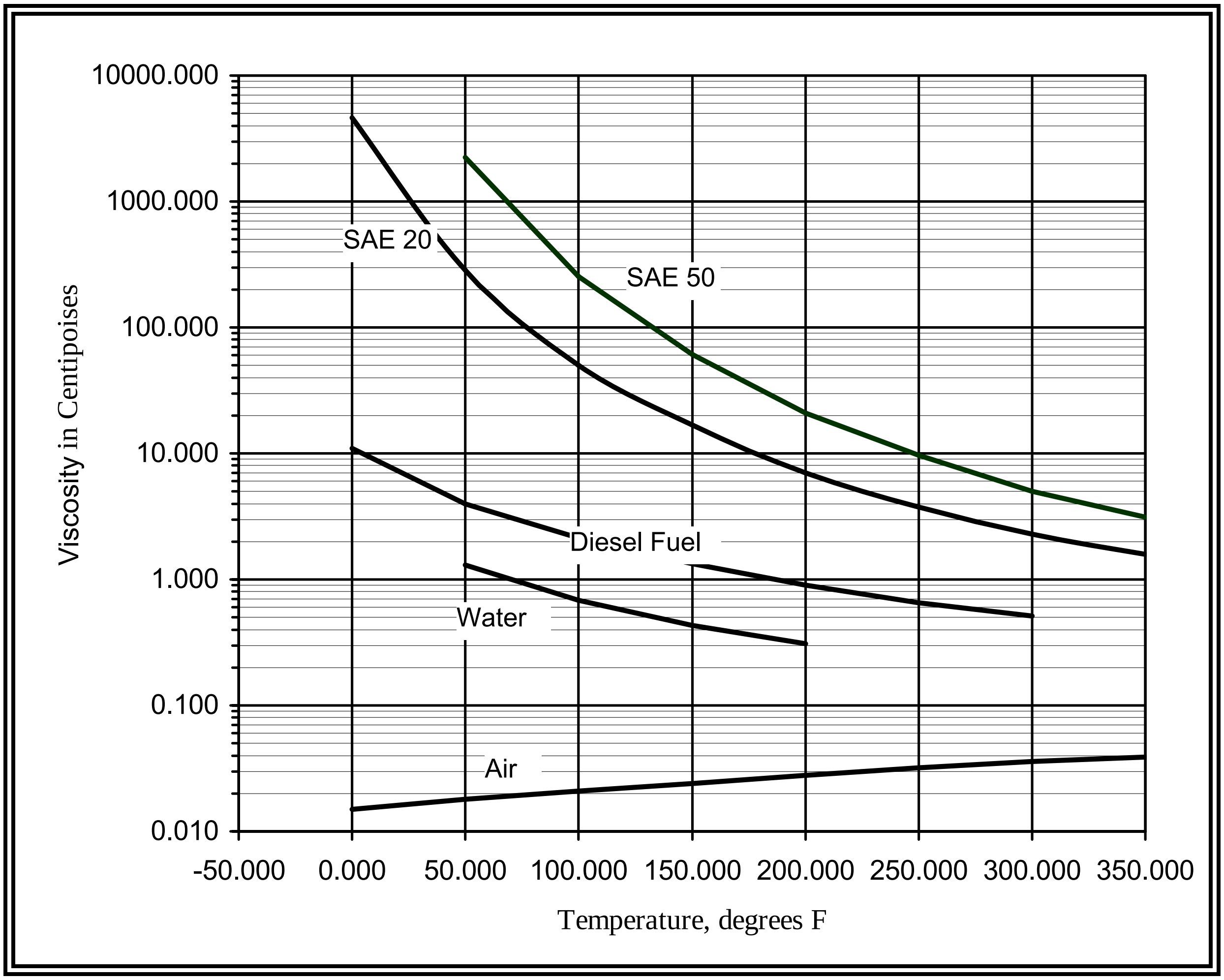


Figure 3.6 Dynamic Viscosities of Various Fluids

3.2.3.6 Fluid Operating Temperature

Operating temperature has a definite effect on the aging process of elastomer and rubber seals. Elevated temperatures, those temperatures above the published acceptable temperature limits, tend to continue the vulcanization or curing process of the materials, thereby significantly changing the original characteristics of the seal or gasket. It can cause increased hardening, brittleness, loss of resilience, cracking, and excessive wear. Since a change in these characteristics has a definite effect on the failure rate of the component, a reliability adjustment must be made.

Temperature will have a significant impact on the performance of a gasket since an increase in temperature will both degrade the physical strength of the material and deform it so that the bolt load and residual stress are modified. Manufacturers of rubber seals will specify the maximum temperature, T_R, for their products. Typical values of T_R are given in Table 3-5. An operating temperature multiplying factor can be derived as follows (Reference 22):

$$C_T = \frac{1}{2^t} \tag{3-11}$$

Where: $t = \dfrac{T_R - T_O}{18}$ for $(T_R - T_O) \leq 40\ ^oF$

T_R = Maximum rated temperature of material, ^oF

T_O = Operating temperature, ^oF

And: C_T = 0.21 for $(T_R - T_O) > 40\ ^oF$

3.2.3.7 Fluid Contaminants

The quantities of contaminants likely to be generated by upstream components are listed in Table 3-4. The number of contaminants depends upon the design, the enclosures surrounding the seal, its physical placement within the system, maintenance practices and quality control. The number of contaminants may have to be estimated from experience with similar system designs and operating conditions.

3.2.3.8 Other Design Analysis Considerations

Those failure rate considerations not specifically included in the model but rather included in the base failure rates are as follows:

- Proper selection of seal materials with appropriate coefficients of thermal expansion for the applicable fluid temperature and compatibility with fluid medium
- Space between the fasteners of a gasket must be small enough so that an even distribution of load is applied to the gasket with fluid pressure
- Potential corrosion from the gland, seal, fluid interface
- Possibility of the seal rolling in its groove when system surges are encountered
- If O-rings can not be installed or replaced easily they are subject to being cut by sharp gland edges
- Potential periods of dryness between applications of fluid

Other factors which need to be considered as a check list for reliability include:

- Chemical compatibility between fluid and seal material
- Thermal stability
- Appropriate thickness and width of the seal material
- Initial and final seating (clamping) force

3.3 DYNAMIC SEALS

In contrast to gaskets and other static seals, dynamic seals are used to control the leakage of fluid in those applications where there is motion between the mating surfaces being sealed. O-rings used in dynamic applications are subject to a sliding action against the gland. This motion introduces friction creating different designs and failure modes from those of static seals. Refer to Section 3.2 for a discussion of seals in general, the basic failure modes of seals and the parameters used in the equations to estimate the failure rate of a seal.

There are several types of dynamic seals including the contacting types such as lip seals and noncontacting types such as labyrinth seals. Assemblies with motion usually require lubrication of the O-ring to reduce wear rate. This is usually accomplished with the fluid being sealed.

Dynamic seals are further divided as follows:

- Reciprocating Seal: A seal where the rod or piston moves back and forth through or with the seal. Piston and rod seals shown in Figure 3.7 are examples of reciprocating seals.

- Rotary Seal: A seal where a shaft rotates with relation to the seal. Typical rotary seals include motor shafts and wheels on a fixed axle. O-rings are not generally used for conditions involving fluid velocities exceeding 800 rpm and/or surface speeds exceeding 600 feet/minute. See Figure 3.7.

- Oscillating Seal: A seal where a shaft turns and returns with relation to the seal. In this application the inner and outer member of the gland moves in an arc around the axis of the shaft first in one direction and then in the opposite direction with the movement usually intermittent. An example application is the faucet valve.

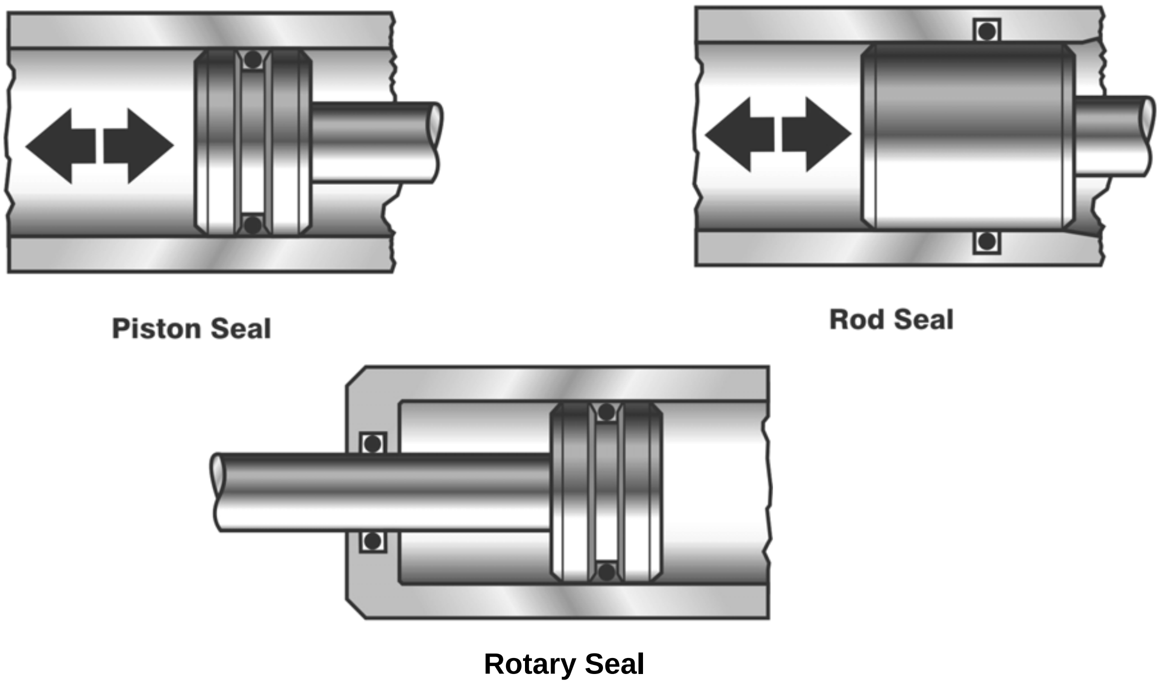


Figure 3.7 Typical Dynamic Seals

The following paragraphs discuss the specific failure modes and model parameters for dynamic seals. Mechanical seals are designed to prevent leakage between a rotating shaft and its housing. The mechanical seal is indicated as the dynamic seal faces in Figure 3.8. Section 3.4 contains specific information on mechanical seals.

3.3.1 Dynamic Seal Failure Modes

The dynamic seal may be used to seal many different liquids at various speeds, pressures, and temperatures. Dynamic seals are made of natural and synthetic rubbers, polymers and elastomers, metallic compounds, and specialty materials. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals. Seals typically operate with sliding contact. Elastomer wear is analogous to metal degradation. However, elastomers are more sensitive to thermal deterioration than to mechanical wear. Hard particles can become embedded in soft elastomeric and metal surfaces leading to abrasion of the harder mating surfaces forming the seal, resulting in leakage.

The most common modes of seal failure are by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals, the wear rate of the seal increasing with the quantity of environmental contamination. A good understanding of the wear mechanism involved will help determine potential seal deterioration. For example, contaminants from the environment such as sand can enter the fluid system and become embedded in the elastomeric seals causing abrasive cutting and damage to shafts.

Compression set refers to the permanent deflection remaining in the seal after complete release of a squeezing load while exposed to a particular temperature level. Compression set reflects the partial loss of elastic memory due to the time effect. Operating over extreme temperatures can result in compression-type seals such as O-rings to leak fluid at low pressures because they have deformed permanently or taken a set after used for a period of time.

Another potential failure mode to be considered is fatigue failure caused by shaft run-out. A bent shaft can cause vibration throughout the equipment and eventual loss of seal resiliency. Typical failure mechanism and causes for dynamic seals are included in Table 3-2.

Table 3-2. Typical Failure Mechanisms and Causes For Dynamic Seals (Also see Table 3-1)

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Excessive leakage	Wear	- Misalignment - Shaft out-of-roundness - Excessive shaft end play - Excessive torque - Poor surface finish - Contaminants - Inadequate lubrication - Excessive rubbing speed
	Dynamic instability	- Shaft misalignment
	Embrittlement	- Contaminants - Fluid/seal incompatibility - Thermal degradation - Idle periods between use
	Mechanical spring Failure	- See Chapter 4, Table 4-1
	Fracture	- Stress-corrosion cracking - Excessive PV value - Excessive fluid pressure on seal
	Edge chipping	- Excessive shaft deflection - Seal faces out-of-square - Excessive shaft whip
	Axial shear	- Excessive pressure loading

Table 3-2. (continued) Typical Failure Mechanisms and Causes For Dynamic Seals (Also see Table 3-1)

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Excessive leakage	Torsional shear	- Excessive torque due to improper lubrication - Excessive fluid pressure
	Compression set	- Extreme temperature operation
	Fluid seepage	- Insufficient seal squeeze - Foreign material on rubbing surface
	Seal face distortion	- Excessive fluid pressure on seal - Foreign material trapped between faces - Excessive PV value of seal operation - Insufficient seal lubrication - Seal shrinkage
Slow mechanical response	Excessive friction	- Excessive squeeze - Excessive seal swell - Seal extrusion - Metal – to – metal contact (out of alignment)

3.3.2 Pressure Velocity

An important factor in the design of dynamic seals is the pressure velocity (PV) coefficient. The PV coefficient is defined as the product of the seal face or system pressure and the fluid velocity. This factor is useful in estimating seal reliability when compared with manufacturer's limits. If the PV limit is exceeded, a seal may wear at a rate greater than desired.

$$Q_S = 0.077 \bullet PV \bullet \mu \bullet a_o \tag{3-12}$$

Where:
- Q_S = Heat input from the seal, BTU/hour
- PV = Pressure-velocity coefficient [See Equation (3-13)]
- μ = Coefficient of friction (See Table 3-6)
- a_o = Seal face area, in^2

The following equation defines the PV factor.

$$PV = \frac{\pi}{12} \bullet DP \bullet d \bullet V \bullet k \quad \textbf{(3-13)}$$

Where:

PV = Pressure-Velocity, $lbs/in^2 \bullet ft/min$

DP = Pressure differential across seal, lbs/in^2

d = Diameter of seal, inches

V = Operating speed, rpm

k = Degree of seal unbalance (1.0 for unbalanced and 0.4 for balanced)

Or:

$$PV = \frac{\pi}{12} \bullet DP \bullet d \bullet V \bullet k$$

Where: V = Operating speed, ft/min

The frictional aspects of materials are not only important from a reliability viewpoint. Performance must also be considered. The more resistance a system incurs, the more power is lost and also the lower the efficiency value for the component.

There should be special consideration for tradeoffs involved with each type of seal material. For example, solid silicon carbide has excellent abrasion resistance, good corrosion resistance, and moderate thermal shock resistance. This material has better qualities than a carbon-graphite base material but has a PV value of 500,000 while carbon-graphite has a 50,000 PV value. With all other values being the same, the heat generated would be five times greater for solid silicon carbide than for carbon-graphite materials. The required cooling flow to the solid silicon carbide seal would be larger to maintain the film thickness on the dynamic seal faces. If this cooling flow can't be maintained, then an increase in wear would occur due to higher surface temperatures. A tradeoff analysis is normally performed for each candidate design to maximize reliability. Typical PV limits are shown in Table 3-7.

3.3.3 Failure Rate Model for Dynamic Seals

Most of the seal modifying factors will remain the same as the ones previously specified by Equation (3-7), the exceptions being surface finish (See Section 3.3.3.1) and the addition of the PV factor for rotational speeds greater than 800 rpm or linear speeds greater than 600 ft/min (See Section 3.3.3.3). The seal model is modified as shown in Equation (3-14).

$$\lambda_{SE} = \lambda_{SE,B} \bullet C_P \bullet C_Q \bullet C_{DL} \bullet C_H \bullet C_F \bullet C_V \bullet C_T \bullet C_N \bullet C_{PV} \tag{3-14}$$

Where:

λ_{SE} = Failure rate of dynamic seal in failures/million hours

$\lambda_{SE,B}$ = Base failure rate of dynamic seal, 22.8 failures/million hours

C_P = Multiplying factor which considers the effect of fluid pressure on the base failure rate for seal movement < 800 rpm or 600ft/min (See Figure 3.10)

C_P = 1.0 for seal movement $\geq$ 800 rpm or 600ft/min

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.11) and Section 3.2.3.2

C_{DL} = Multiplying factor which considers the effect of seal size on the base failure rate (See Figure 3.12) for seal movement < 800 rpm or 600ft/min

C_{DL} = 1.0 for seal movement $\geq$ 800 rpm or 600 ft/min

C_H = Multiplying factor which considers the effect of contact stress and seal hardness on the base failure rate for movement < 800 rpm or 600 ft/min (See Figure 3.14)

C_H = 1.0 for seal movement $\geq$ 800 rpm or 600 ft/min

C_F = Multiplying factor which considers the effect of surface finish on the base failure rate (See Sections 3.3.3.1, 3.2.3.4 and Figure 3.17)

C_V = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3 and Section 3.2.3.5)

C_T = Multiplying factor which considers the effect of seal temperature on the base failure rate (See Figure 3.16 and Section 3.2.3.6)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4 and Section 3.3.3.2)

C_{PV} = Multiplying factor which considers the effect of the pressure-velocity coefficient on the base failure rate for movement ≥ 800 rpm and/or 600 ft/min (See Sections 3.3.2 and 3.3.3.3)

C_{PV} = 1.0 for seal movement < 800 rpm or 600 ft/min

3.3.3.1 Surface Finish Multiplying Factor

Surface irregularities of dynamic seals may be more pronounced than static seals. In dynamic seal applications where the seal mates with a shaft, shaft hardness, smoothness and material are factors which must be considered in the design evaluation process. Maximum seal efficiency and life are obtained with a finely finished gland surface, usually in the 10 to 20 microinch range. A metal surface finish of less than 8 microinches rms increases the total frictional drag of a compound moving against it. The degree to which the finish can be maintained in the operating range must be considered when determining the surface finish of the gland for use in the model. Figure 3.17 provides a value for the surface finish multiplying factor as a function of the surface finish.

3.3.3.2 Fluid Contaminant Multiplying Factor

One of the factors in estimating the failure rate of a dynamic seal is the number of contaminants in contact with the seal generated from other components in the system. For example, when a cylinder rod extends out into a dirty environment where it can pick up dirt, lint, metal chips and other contaminants, this foreign material can nullify the benefits of the lubricant and cause rapid abrasive wear of both the O-ring and the rod. Equipment exposed to such conditions should contain a wiper ring to prevent the foreign material from reaching the O-ring. A felt ring is usually installed between the wiper and the seal to maintain lubrication of the rod during its return stroke.

Table 3-4 provides fluid contaminant multiplying factors for various components that may be generating contaminants.

3.3.3.3 PV Multiplying Factor

C_{PV} is the multiplying factor that multiplies the base failure rate by the ratio of PV value for actual seal operation to design PV value. C_{PV} is applicable to rotary seals, lip seals and other dynamic seals that rotate with a shaft or reciprocate with a velocity greater than 5 in/sec and where a PV design factor is available from the manufacturer. The values for PV_{DS} and PV_{OP} used in Equation (3-15) will use the PV formulation in Equation (3-13).

$$C_{PV} = \frac{PV_{OP}}{PV_{DS}} \quad \textbf{(3-15)}$$

Where: PV_{OP} = PV factor for actual seal operation

PV_{DS} = PV factor for the original design

3.4 MECHANICAL SEALS

Mechanical seals are designed to prevent leakage between flat, rotating surfaces. A typical contacting type dynamic seal is shown in Figure 3.8. In this example, the sealing surfaces are perpendicular to the shaft, with contact between the primary and mating rings to achieve a dynamic seal. Each of the sealing surfaces is lapped flat to eliminate leakage. Wear occurs at the dynamic seal faces from sliding contact between the primary and mating rings. The rate of wear is small, as a film of the liquid sealed is maintained between the sealing faces. Preload from a spring is required to produce an initial seal, the spring pressure holding the primary and mating rings together during shutdown or when there is a lack of fluid pressure.

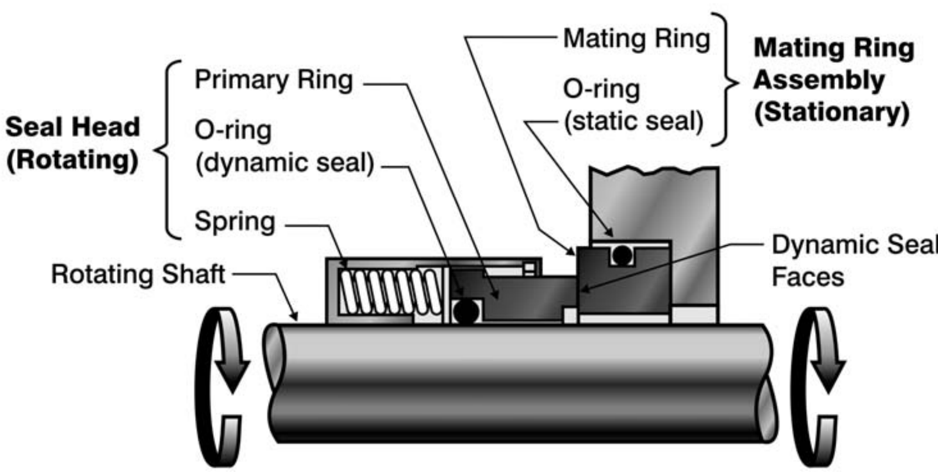


Figure 3.8 Typical Mechanical Seal

Wear occurs between the primary ring and mating ring of a mechanical seal. This surface contact is maintained by a spring. There is a film of liquid maintained between the sealing surfaces to eliminate as much friction as possible. For most mechanical seals, the three common points of sealing contact occur between the following points:

(1) Mating surfaces between primary and mating rings

(2) Between the rotating component and shaft or sleeve

(3) Between the stationary component and the gland plate

The reliability of a mechanical seal depends to a very large extent on its ability to maintain a thin fluid film in the gap between the mating faces while simultaneously minimizing the duration and extent of mechanical contact between asperities on the rubbing areas of these faces. Too much contact may overheat the materials; not enough contact may cause high leakage rates.

A mechanical seal may be an unbalanced design or a balanced design. Unbalanced seals are seal arrangements in which the hydraulic pressure of the seal chamber acts on the entire seal face without any of the force being reduced through the seal design. Unbalanced seals usually have a lower pressure limitation than balanced seals. A balanced seal design reduces the hydraulic forces acting on the seal faces through mechanical seal design. As the seal faces rub together, the amount of heat generated is determined by the amount of pressure applied, the lubricating film between the faces, the rotational speed, and the seal ring materials. Balanced seals reduce the seal ring area on which the stuffing box pressure acts. With a reduction in area, the overall closing force is diminished. This results in better lubrication and reduced heat generation and face wear compared to unbalanced seals. Unbalanced and balanced seal designs are shown in Figure 3.9. The failure rate equation assumes a balanced seal.

3.4.1 Mechanical Seal Failure Modes

Failure modes of a mechanical seal can be identified by three main causes of failure: temperature, pressure and velocity and a combination of these variables. For example, fluid pressure can create extra heat at the seal face which in turn can increase the rate of wear and other destructive failure modes such as material fracture and distortion and leakage. Elastomer seals can become extruded and damaged. As the pressure is increased, the probability of failure goes up.

Some mechanical seals wear out with use and some fail prior to wearing out. The seal face is the only part of a mechanical seal designed to wear out. Mechanical face seals should last until the carbon face wears away. If the seal starts leaking before that happens and the seal requires replacement, then the seal has failed. In some cases the seal face has opened because it became jammed on the rotating component. Another possibility is that one of the seal components such as the spring was damaged by contact, heat or corrosion.

Wear often occurs between the primary ring and mating ring. This surface contact is maintained by a spring. There is a film of liquid maintained between the sealing surfaces to eliminate as much friction as possible. For most mechanical seals the three common points of sealing contact occur between the following points:

(1) Mating surfaces between primary and mating rings

(2) Between the rotating component and shaft or sleeve

(3) Between the stationary component and the gland plate

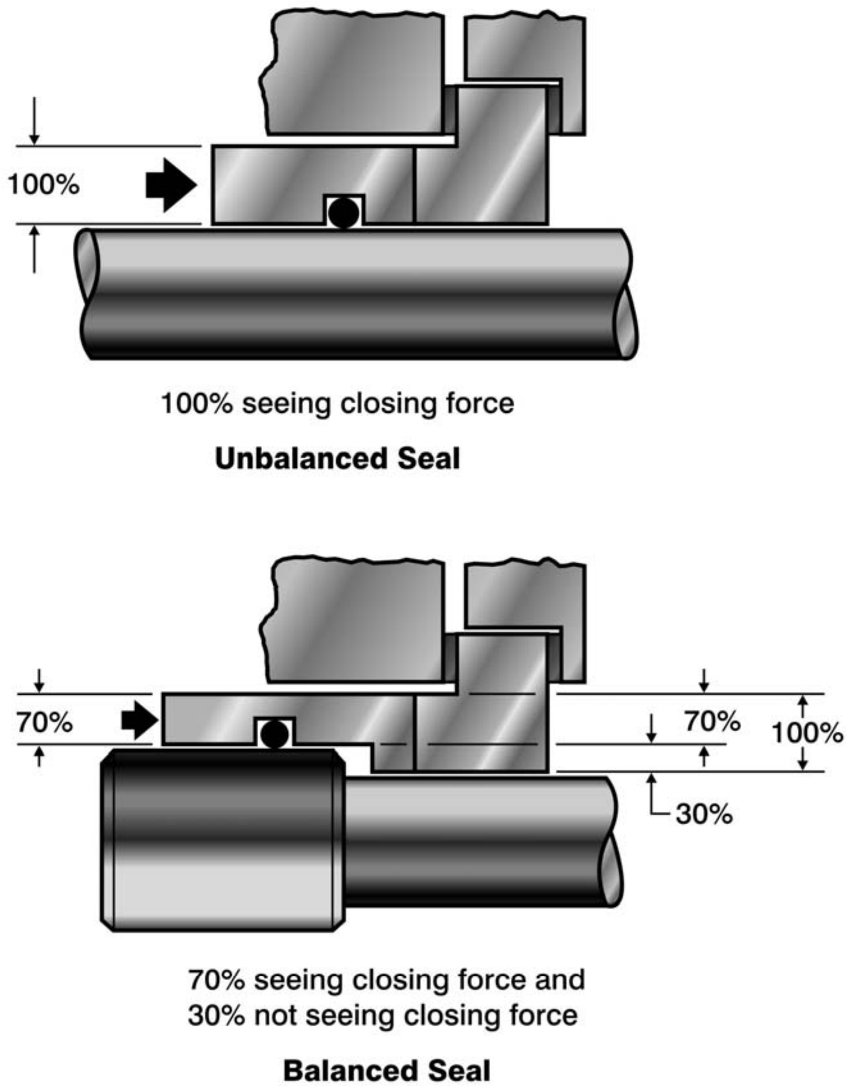


Figure 3.9 Balanced and Unbalanced Seal Designs

Seal balance is a performance characteristic that measures how effective the seal mating surfaces match. If not effectively matched, the seal load at the dynamic facing may be too high causing the liquid film to be squeezed out and vaporized, thus causing a high wear rate. The fluid pressure from one side of the primary ring causes a certain amount of force to impinge on the dynamic seal face. The dynamic facing pressure can be controlled by manipulating the hydraulic closing area with a shoulder on a sleeve or by seal hardware. By increasing the area, the sealing force is increased.

The reliability of a mechanical seal depends to a very large extent on its ability to maintain a thin film in the gap between the mating surfaces and at the same time minimizing the mechanical contact of the face surfaces. Too much contact may cause overheating of the face materials and insufficient contact may cause excessive leakage. Over time common modes of seal failure are by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals, the wear rate of the seal increasing with the quantity of environmental contamination. Contaminants from the environment such as sand can enter the fluid system and become embedded in the elastomeric seals causing abrasive cutting and damage to shafts as well as the mechanical seal.

There should be special consideration for tradeoffs involved with each type of seal material. For example, solid silicon carbide has excellent abrasion resistance, good corrosion resistance, and moderate thermal shock resistance. This material has better qualities than a carbon-graphite base material but has a *PV* value of 500,000 lb/in-min while carbon-graphite has a 50,000 lb/in-min *PV* value. With all other values being the same, the heat generated would be five times greater for solid silicon carbide than for carbon-graphite materials. The required cooling flow to the solid silicon carbide seal would be larger to maintain the film thickness on the dynamic seal faces. If this cooling flow can't be maintained, then an increase in wear would occur due to higher surface temperatures.

3.4.2 Failure Rate Model for Mechanical Seals

An important seal design consideration is seal balance. Seal balance refers to the difference between the pressure of the fluid being sealed and the contact pressure between the seal faces. It is the ratio of hydraulic closing area to seal face area (parameter k in Equation (3-13). A balanced seal is designed so that the effective contact pressure is always less than the fluid pressure, reducing friction at the seal faces. The result is less rubbing wear, less heat generated and higher fluid pressure capability. In an unbalanced seal, fluid pressure is not relieved by the face geometry, the seal faces withstand full system fluid pressure in addition to spring pressure and the face contact pressure is greater than or equal to fluid pressure. The failure rate equation assumes a balanced seal.

Of greatest importance with mechanical seals is a properly designed seal face. Proper mating surface materials must be matched so that excessive heat isn't generated from the dynamic motion of the seal faces. Too much heat can cause thermal distortions on the face of the seal and cause gaps which can increase the leakage rate. It can also cause material changes that can significantly increase the seal wear rate. Therefore, a careful review of the seal material should be made for each surface of the dynamic seal face. Equation (3-12) (Reference 26) includes such coefficients of friction and wear rate. Table 3-6 shows frictional values for various seal face materials.

An important factor in the design of mechanical seals is the pressure velocity (PV) coefficient. The PV coefficient is defined as the product of the seal face or system pressure and the fluid velocity. This factor is useful in estimating seal reliability when compared with manufacturer's limits. If the PV limit is exceeded, a seal may wear at a rate greater than desired. PV limits are included in manufacturer's specification sheets.

$$\lambda_{SE} = \lambda_{SE,B} \bullet C_Q \bullet C_F \bullet C_V \bullet C_T \bullet C_N \bullet C_{PV} \tag{3-16}$$

Where:

λ_{SE} = Failure rate of mechanical seal in failures/million hours

$\lambda_{SE,B}$ = Base failure rate of mechanical seal, 22.8 failures/million hours

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.11) and Section 3.2.3.2

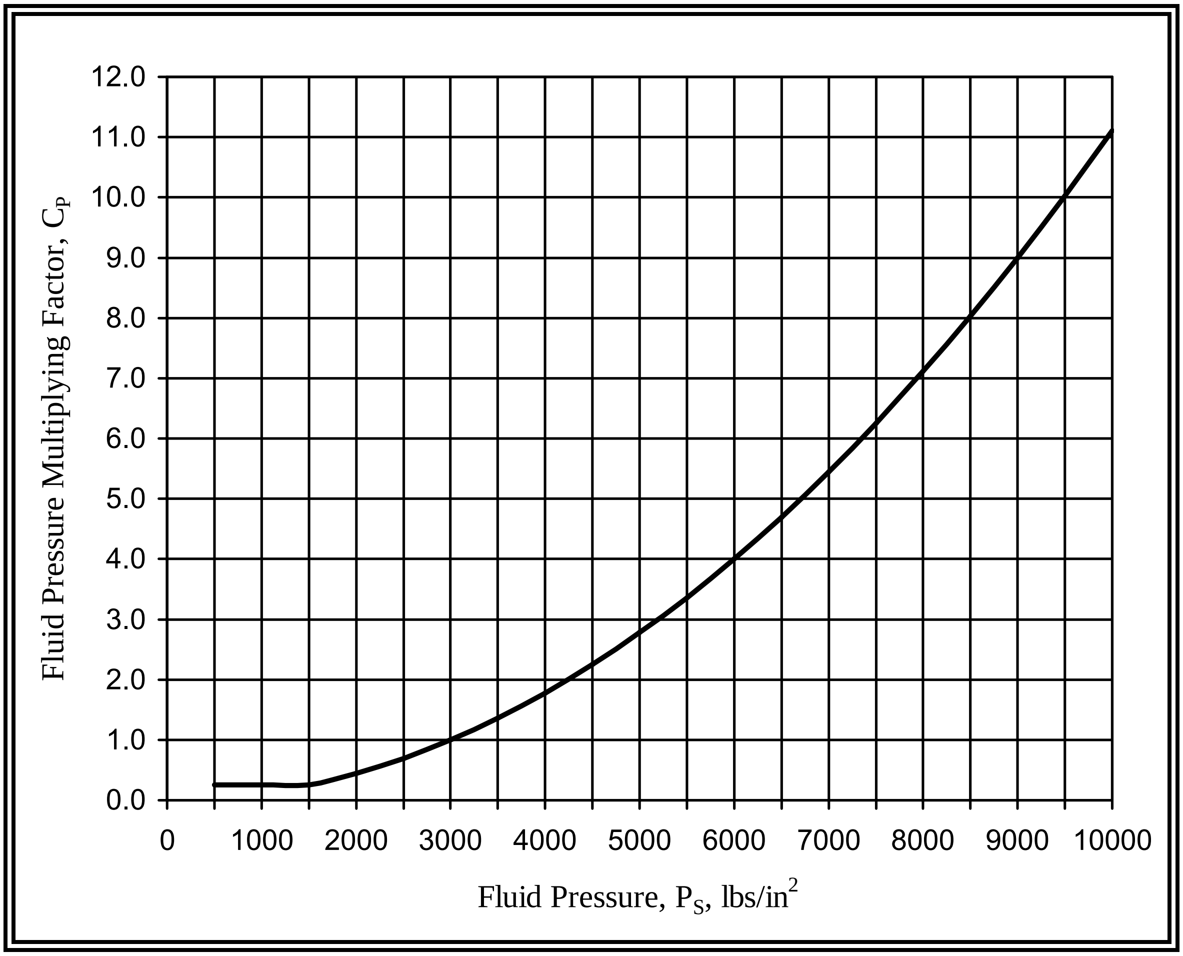
C_f = Multiplying factor which considers the effect of seal face surface finish on the base failure rate (See Sections 3.3.3.1, 3.2.3.4 and Figure 3.17)

C_V = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3 and Section 3.2.3.5)

C_T = Multiplying factor which considers the effect of seal face temperature on the base failure rate (See Figure 3.16 and Section 3.2.3.6)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4 and Section 3.3.3.2)

C_{PV} = Multiplying factor which considers the effect of the pressure-velocity coefficient on the base failure rate (See Sections 3.3.2 and 3.3.3.3)

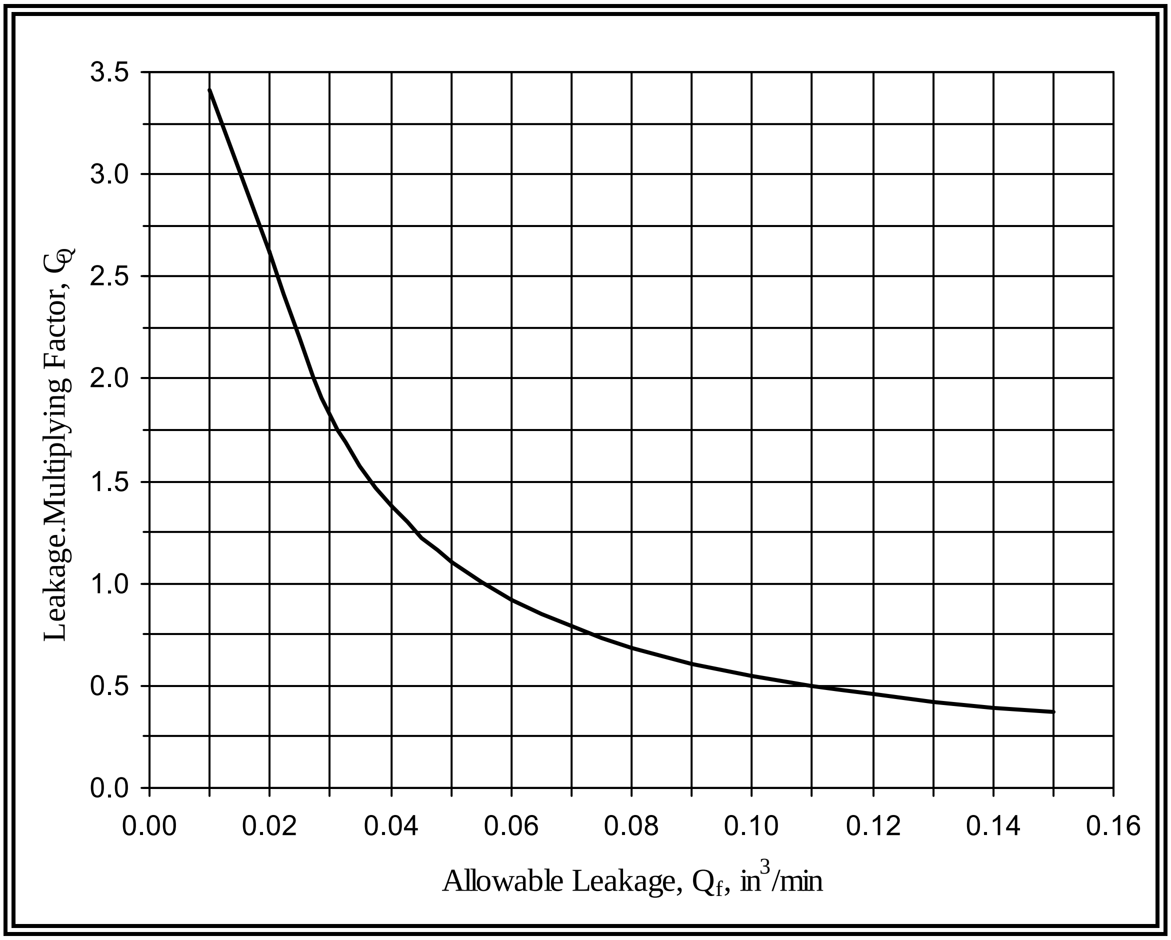


For $P_S \le 1500$ lbs/in^2, $C_P = 0.25$

For $P_S > 1500$ lbs/in^2, $C_P = \left(\frac{P_S}{3000}\right)^2$

Where $P_S = P_1 - P_2$ (upstream – downstream pressure)

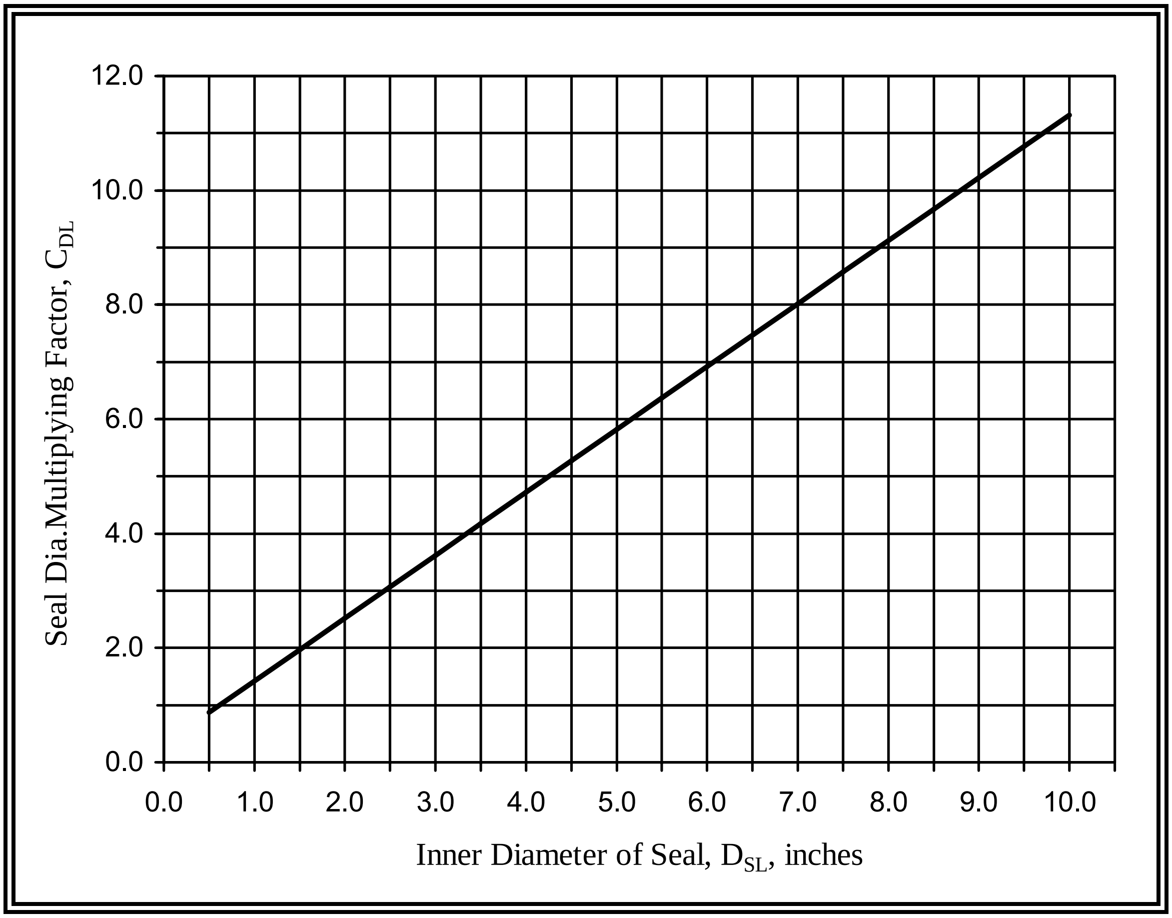
Figure 3.10 Fluid Pressure Multiplying Factor, C_P



For Leakage $>$ 0.03 in^3/min, $C_Q = 0.055/Q_f$

For Leakage $\leq$ 0.03 in^3/min, $C_Q = 4.2 - (79\ Q_f)$

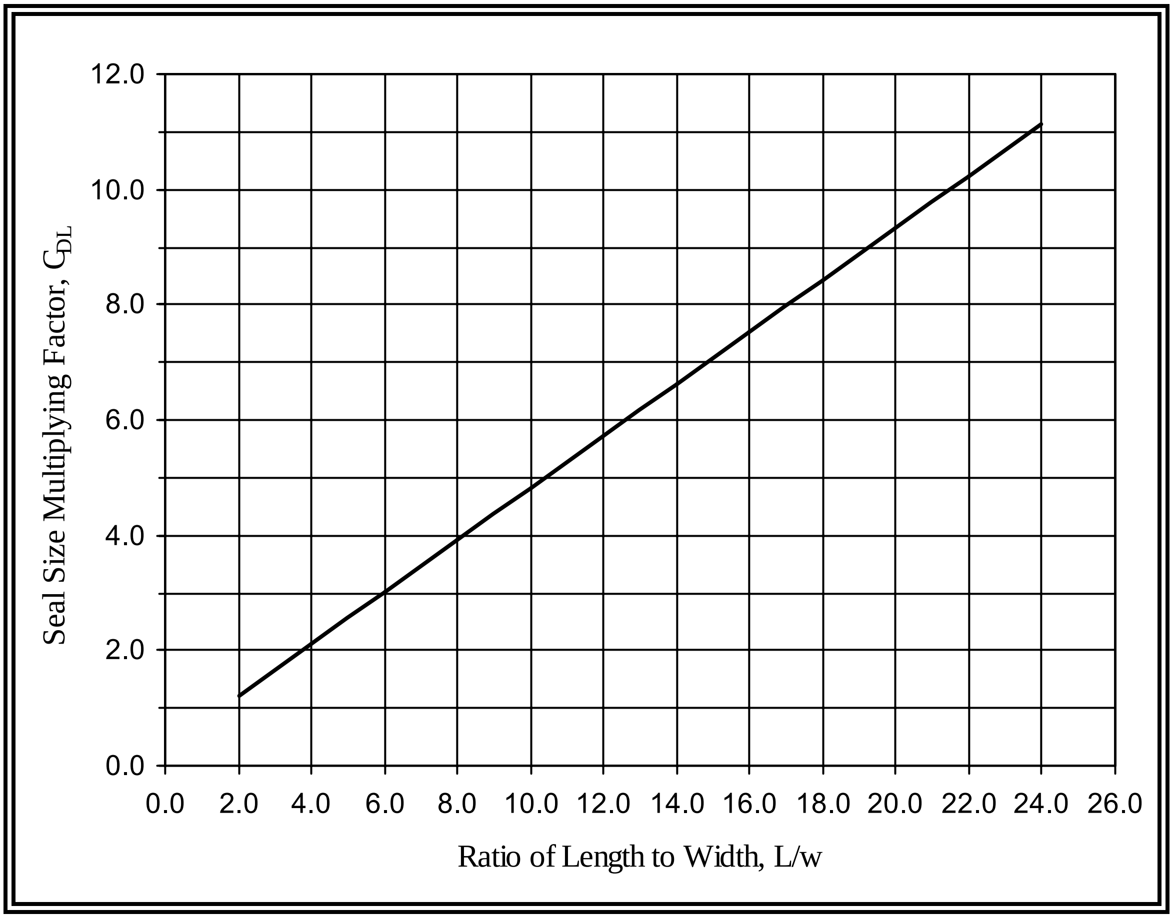
Figure 3.11 Allowable Leakage Multiplying Factor, C_Q



$$C_{DL} = 1.1\, D_{SL} + 0.32$$

Where: D_{SL} = Inner diameter of seal

Figure 3.12 Seal Diameter Multiplying Factor, C_{DL}

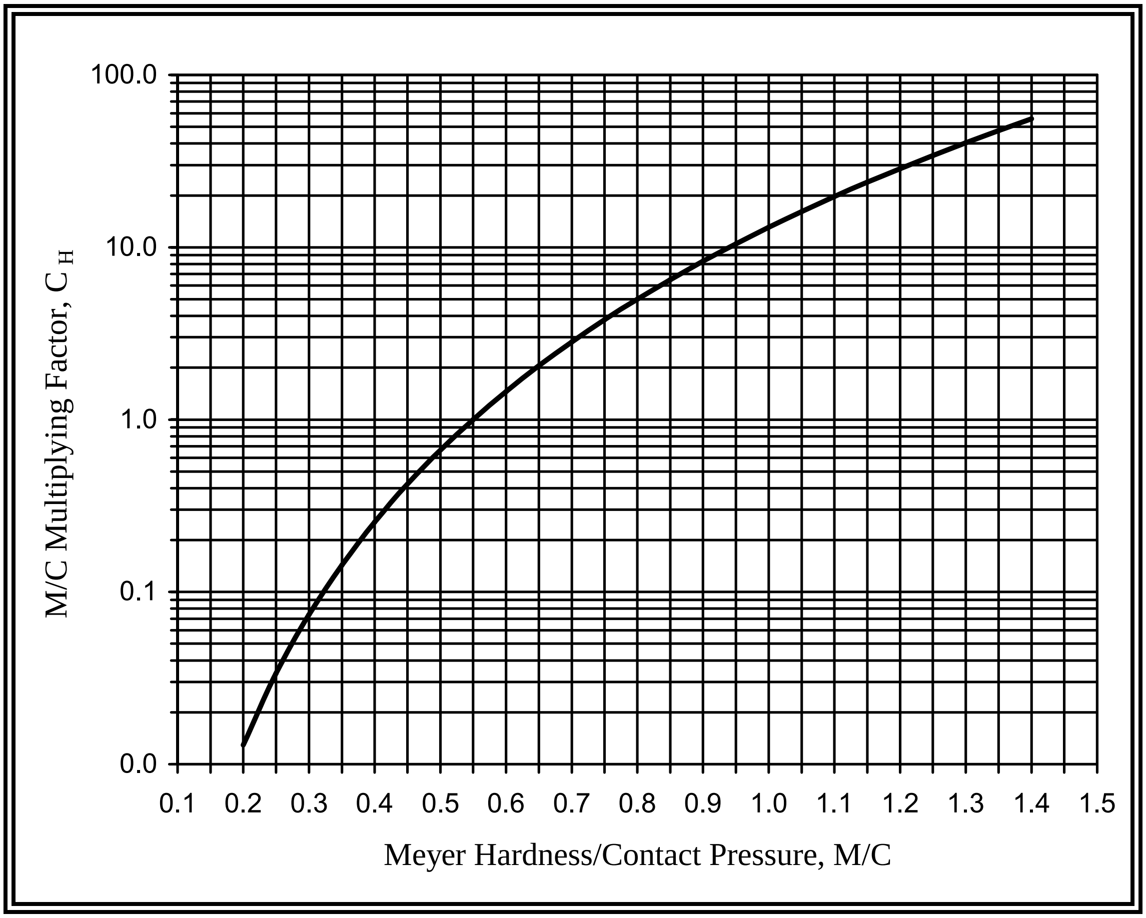


$$C_{DL} = 0.45\left(\frac{L}{w}\right) + 0.32$$

Where: L = Total linear length of gasket

w = Minimum width of gasket

Figure 3.13 Gasket Size Multiplying Factor, C_{DL}

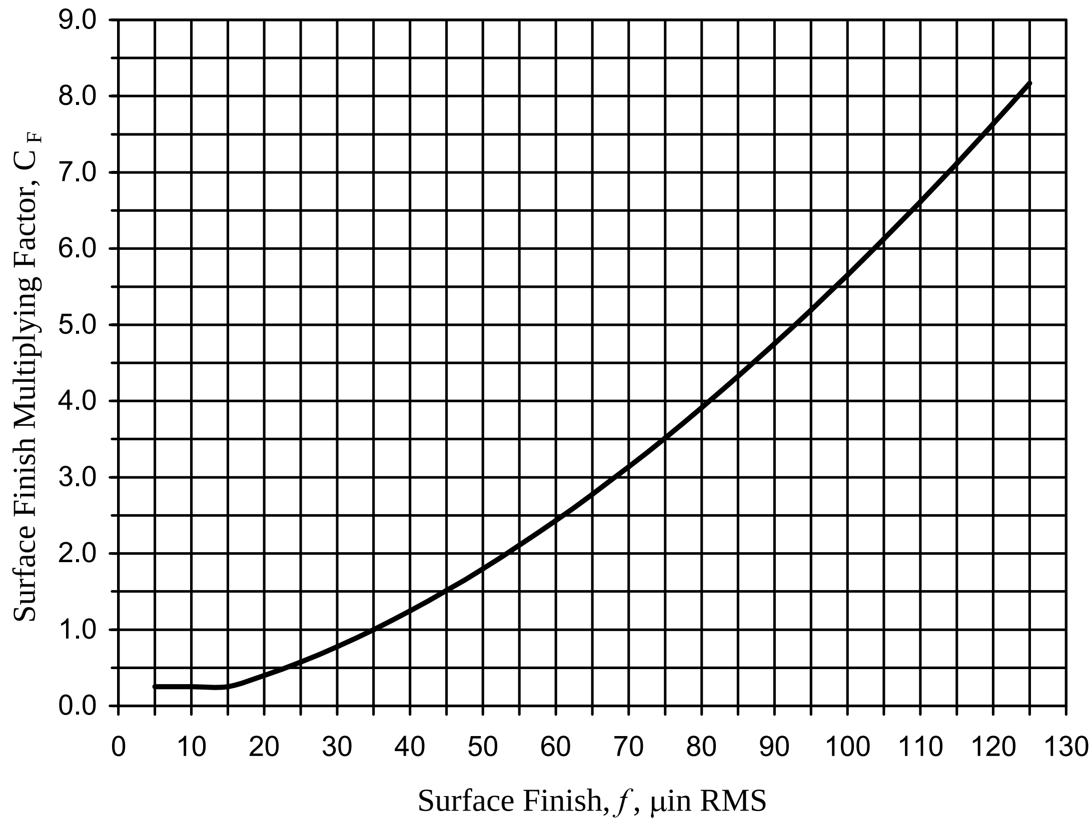


$$C_H = \left(\frac{M / C}{0.55} \right)^{4.3}$$

Where: M = Meyer Hardness, lbs/in^2

C = Contact Pressure, lbs/in^2

Figure 3.14 Material Hardness/Contact Pressure, C_H

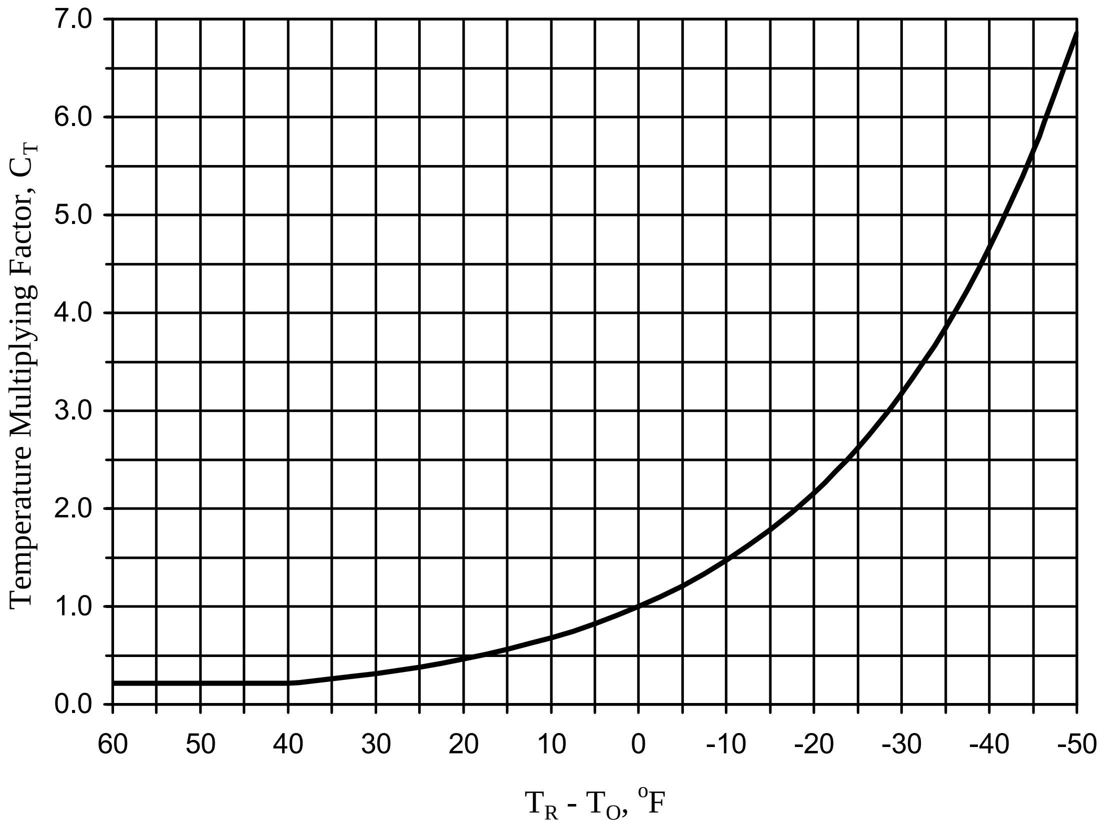


For $f \leq$ 15 µin, $C_f = 0.25$

For f > 15 µin, $C_f = \dfrac{f^{1.65}}{353}$

Where: f = Surface Finish, µin RMS

Figure 3.15 Surface Finish Multiplying Factor, C_F (for static seals)



$$C_T = \frac{1}{2^t}$$

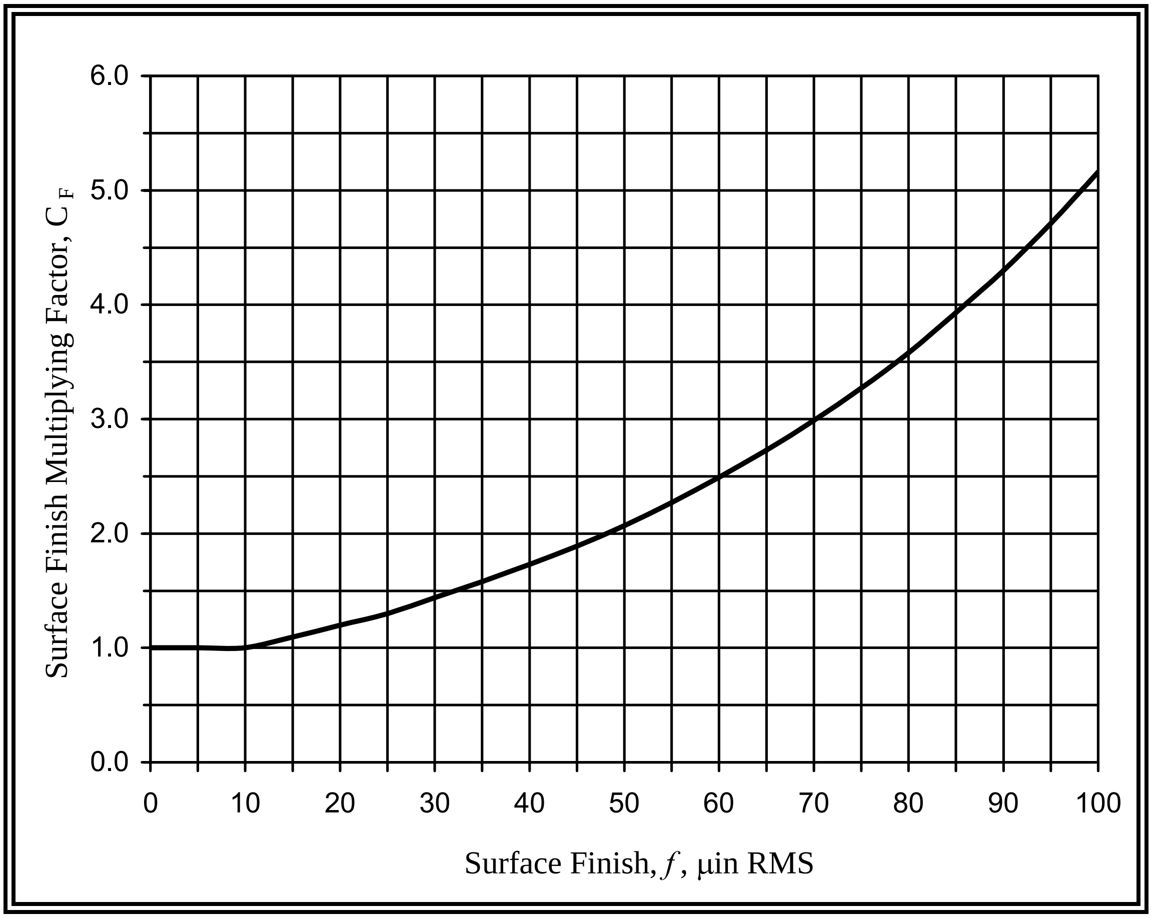
Where: $t = \frac{(T_R - T_O)}{18}$ for $(T_R - T_O) \leq 40\ ^oF$

and: $C_T = 0.21$ for $(T_R - T_O) > 40\ ^oF$

T_R = Rated Temperature of Seal, ^oF (See Table 3-6)

T_O = Operating Temperature of Seal, ^oF

Figure 3.16 Temperature Multiplying Factor, C_T



For $f \leq 10$ µin, $C_f = 1.00$

For $f > 10$ µin, $C_f = \dfrac{1}{2^{((10-f)/38)}}$

Where: f = Surface Finish, µin RMS

Figure 3.17 Surface Finish Multiplying Factor, C_F (for dynamic seals)

Table 3-3. Fluid Viscosity/Temperature Multiplying Factor, C_ν for Typical Fluids

FLUID	C_ν								
	Fluid Temperature, ^oF								
	-50	0	50	100	150	200	250	300	350
Air	554.0	503.4	462.9	430.1	402.6	379.4	359.5	---	---
Oxygen	504.6	457.8	420.6	390.2	365.9	343.6	325.3	---	---
Nitrogen	580.0	528.0	486.5	452.6	424.3	400.0	379.6	---	---
Carbon Dioxide	---	599.9	510.7	449.7	395.9	352.1	---	---	---
Water	---	---	6.309	12.15	19.43	27.30	---	---	---
SAE 10 Oil	---	---	0.060	0.250	0.750	1.690	2.650	---	---
SAE 20 Oil	---	---	0.0314	0.167	0.492	1.183	2.213	2.861	5.204
SAE 30 Oil	---	---	0.0297	0.1129	0.3519	0.8511	1.768	2.861	4.309
SAE 40 Oil	---	---	0.0122	0.0534	0.2462	0.6718	1.325	2.221	3.387
SAE 50 Oil	---	---	0.0037	0.0326	0.1251	0.3986	0.8509	1.657	2.654
SAE 90 Oil	---	---	0.0012	0.0189	0.0973	0.3322	0.7855	1.515	2.591
Diesel Fuel	0.1617	0.7492	2.089	3.847	6.228	9.169	12.78	16.31	---
MIL-H-83282	0.0031	0.0432	0.2137	0.6643	1.421	2.585	4.063	0.6114	0.7766
MIL-H-5606	0.0188	0.0951	0.2829	0.6228	1.108	1.783	2.719	3.628	4.880

--- Data for these temperatures determined to be unreliable

$$C_\nu = \left(\frac{\nu_O}{\nu}\right)$$

Where: ν_O = 2 x 10^{-8} lbf-min/in^2

ν = Dynamic viscosity of fluid being used, lbf-min/in^2

Table 3-4. Contaminant Multiplying Factor, C_N

TYPICAL QUANTITIES OF PARTICLES PRODUCED BY HYDRAULIC COMPONENTS	PARTICLE MATERIAL	NUMBER PARTICLES UNDER 10 MICRON PER HOUR PER RATED GPM (N10)
Piston Pump	steel	0.017
Gear Pump	steel	0.019
Vane Pump	steel	0.006
Cylinder	steel	0.008
Sliding action valve	steel	0.0004
Hose	rubber	0.0013

$$C_N = \left(\frac{C_O}{C_{10}}\right)^3 \bullet FR \bullet N_{10}$$

Where:
C_o = System filter size in microns
C_{10} = Standard system filter size = 10 micron
FR = Rated flow rate, GPM
N_{10} = Particle size factor

Table 3-5. T_R Values for Typical Seal Materials (Reference 27)

SEAL MATERIAL	T_R (°F)
Natural rubber	160
Ethylene propylene	250
Neoprene	250
Nitrile	250
Polyacrylate	300
Fluorosilicon	450
Fluorocarbon	475
Silicon rubbers	450
Butyl rubber	250
Urethane	210
Fluroelastomers	500
Fluroplastics	500
Leather	200
Impregnated poromeric material	250

Table 3-6. Coefficient of Friction for Various Seal Face Materials

SLIDING MATERIALS		COEFFICIENT OF FRICTION (μ)
ROTATING (seal head)	**STATIONARY (mating ring)**	
Carbon-graphite (resin filled)	- Cast Iron	0.07
	- Ceramic	0.07
	- Tungsten Carbide	0.07
	- Silicon Carbide	0.02
	- Silicon Carbide Converted Carbon	0.015
Silicon carbide	- Tungsten Carbide	0.02
	- Silicon Carbide Converted Carbon	0.05
	- Silicon Carbide	0.02
	- Tungsten Carbide	0.08

Table 3-7. Typical Pressure Velocity (PV) Limits

Face Materials	PV (lb/in^2 ft/min)
Carbon vs hard faced stainless steel	543,000
Carbon vs ceramic	543,000
Carbon vs leaded bronze	992,000
Carbon vs nickel iron	1,142,000
Carbon vs tungsten carbide	2,570,000

3.5 REFERENCES

In addition to specific references cited throughout Chapter 3, other references included below are recommended in support of performing a reliability analysis of seals and gaskets.

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